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SECTION 5

CLUTCHES

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Basic Types of Mechanical Clutches

Sketches include both friction and positive types. Figs. 1-7 are classified as externally controlled; Figs. 8-12 are internally controlled. The latter are further divided into overload relief, over-riding, and centrifugal types.

Marvin Taylor

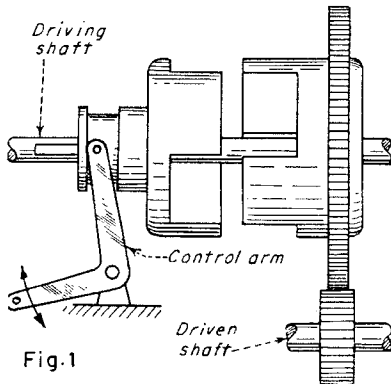


Fig. 1

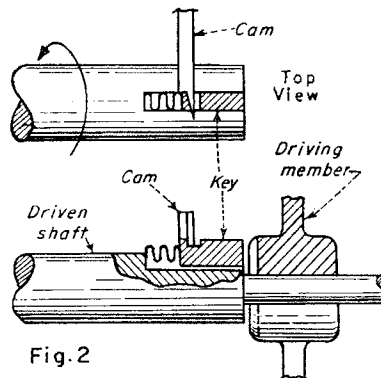


Fig. 2

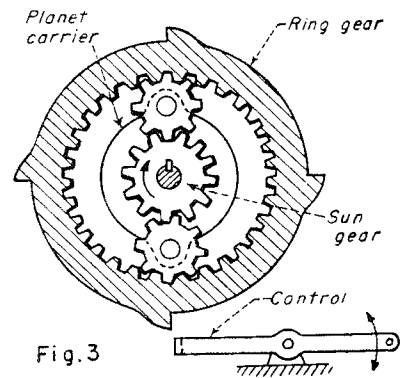


Fig. 3

1. JAW CLUTCH. Left sliding half is feathered to the driving shaft while right half rotates freely. Control arm activates the sliding half to engage or disengage the drive. This clutch, though strong and simple, suffers from disadvantages of high shock during engagement, high inertia of the sliding half, and considerable axial motion required for engagement.

2. SLIDING KEY CLUTCH. Driven shaft with a key-way carries freely-rotating member which has radial slots along its hub; sliding key is spring loaded but is normally restrained from engaging slots by the control cam. To engage the clutch, control cam is raised and key

enters one of the slots. To disengage, cam is lowered into the path of the key; rotation of driven shaft forces key out of slot in driving member. Step on control cam limits axial movement of the key.

3. PLANETARY TRANSMISSION CLUTCH. In disengaged position shown, driving sun gear will merely cause the free-wheeling ring gear to idle counter-clockwise, while the driven member, the planet carrier, remains motionless. If motion of the ring gear is blocked by the control arm, a positive clockwise drive is established to the driven planet carrier.

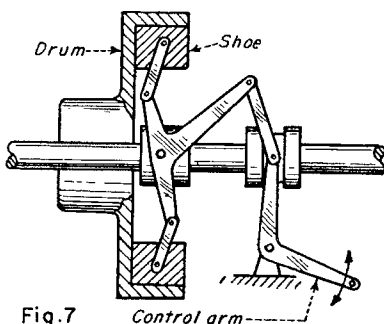


Fig. 7

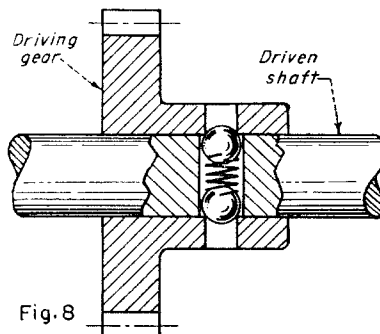


Fig. 8

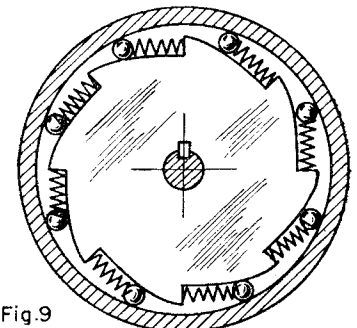


Fig. 9

7. EXPANDING SHOE CLUTCH. In sketch above, engagement is obtained by motion of control arm which operates linkages to force friction shoes radially outward into contact with inside surface of drum.

8. SPRING AND BALL RADIAL DETENT CLUTCH. This design will positively hold the driving gear and driven shaft in a given timing relationship until the torque becomes excessive. At this point the balls will be forced inward against their spring pressure and out of engagement with the holes in the hub, thus permitting the driving gear to continue rotating while the driven shaft is stationary.

9. CAM AND ROLLER CLUTCH. This over-running clutch is suited for higher speed free-wheeling than the pawl and ratchet types. The inner driving member has camming surfaces at its outer rim and carries light springs that force rollers to wedge between these surfaces and the inner cylindrical face of the driven member. During driving, self-energizing friction rather than the springs forces the roller to tightly wedge between the members and give essentially positive drive in a clockwise direction. The springs insure fast clutching action. If the driven member should attempt to run ahead of the driver, friction will force the rollers out of a tight wedging position and break the connection.

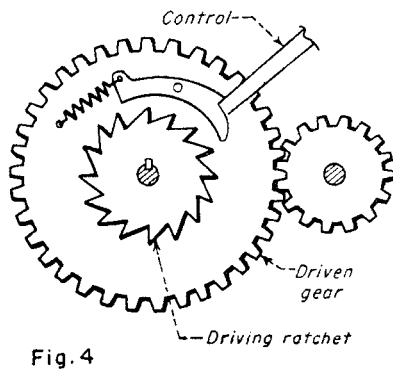


Fig. 4

4. PAWL AND RATCHET CLUTCH. (External Control). Ratchet is keyed to the driving shaft; pawl is carried by driven gear which rotates freely on the driving shaft. Raising the control member permits the spring to pull the pawl into engagement with the ratchet and drive the gear. Engagement continues until control member is lowered into the path of a camming surface on the pawl. The motion of the driven gear will then force the pawl out of engagement and bring the driven assembly to a solid stop against the control member. This clutch can be converted into an internally controlled type of unit by removing the external control arm and replacing it with a slideable member on the driving shaft.

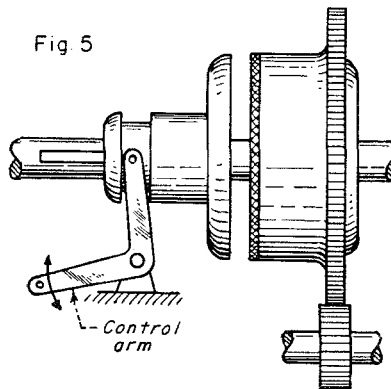


Fig 5

5. PLATE CLUTCH. Available in many variations, with single and multiple plates, this unit transmits power through friction force developed between the faces of the left sliding half which is fitted with a feather key and the right half which is free to rotate on the shaft. Torque capacity depends upon the axial force exerted by the control member when it activates the sliding half.

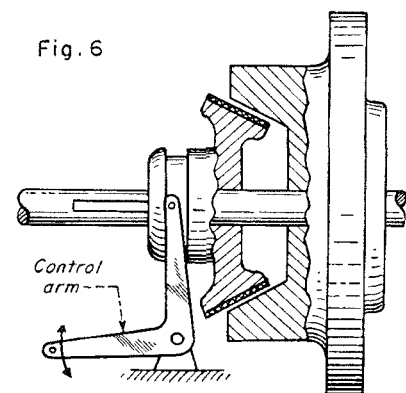


Fig. 6

6. CONE CLUTCH. This type also requires axial movement for engagement, but the axial force required is less than that required with plate clutches. Friction material is usually applied to only one of the mating surfaces. Free member is mounted to resist axial thrust.

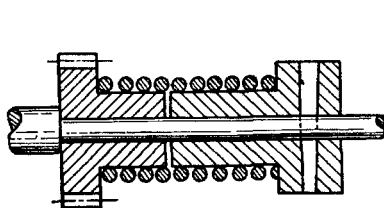


Fig. 10

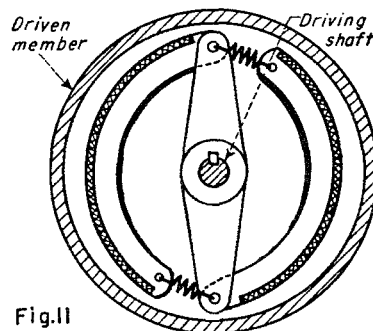


Fig. 11

10. WRAPPED SPRING CLUTCH. Makes a simple and inexpensive uni-directional clutch consisting of two rotating members connected by a coil spring which fits snugly over both hubs. In the driving direction the spring tightens about the hubs producing a self energizing friction grip; in the opposite direction it unwinds and will slip.

11. EXPANDING SHOE CENTRIFUGAL CLUTCH. Similar in action to the unit shown in Fig. 7 with the exception that no external control is used. Two friction shoes, attached to the driving member, are held inward by springs until they reach the "clutch-in" speed, at

which centrifugal force energizes the shoes outward into contact with the drum. As the driver rotates faster the pressure between the shoes and the drum increases thereby providing greater torque capacity.

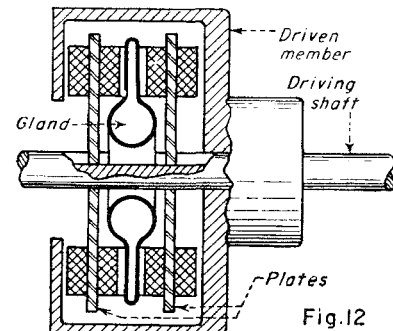
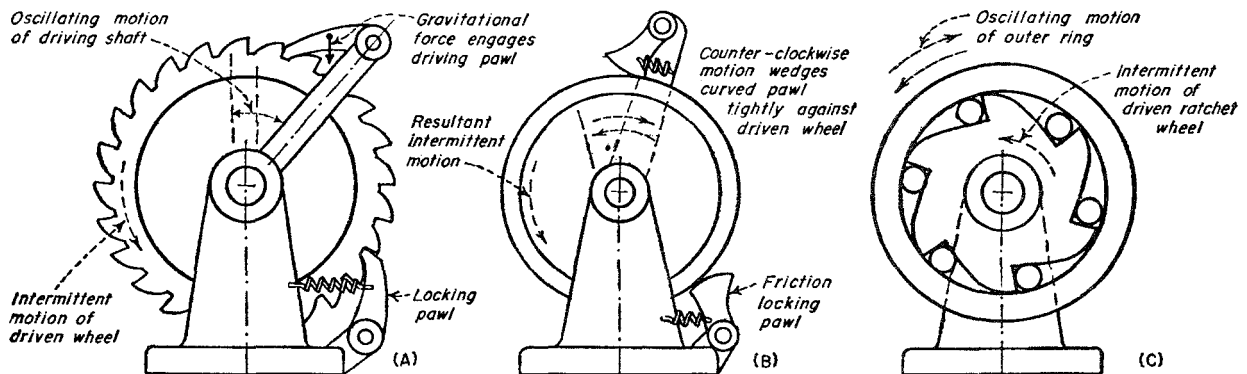


Fig. 12

12. MERCURY GLAND CLUTCH. Contains two friction plates and a mercury filled rubber gland, all keyed to the driving shaft. At rest, mercury fills a ring shaped cavity near the shaft; when revolved at sufficient speed, the mercury is forced outward by centrifugal force spreading the rubber gland axially and forcing the friction plates into driving gland contact with the faces of the driven housing. Axial thrust on driven member is negligible.

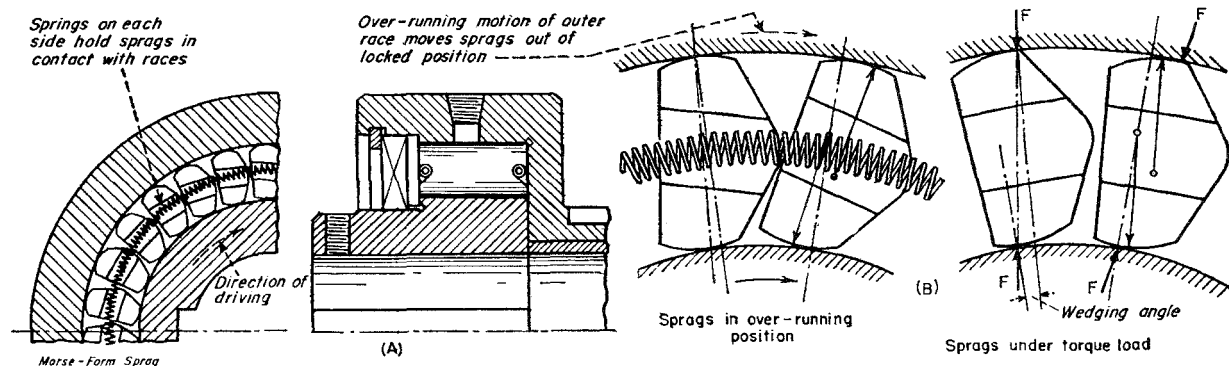
Construction Details of Over-Riding Clutches

A. DeFeo



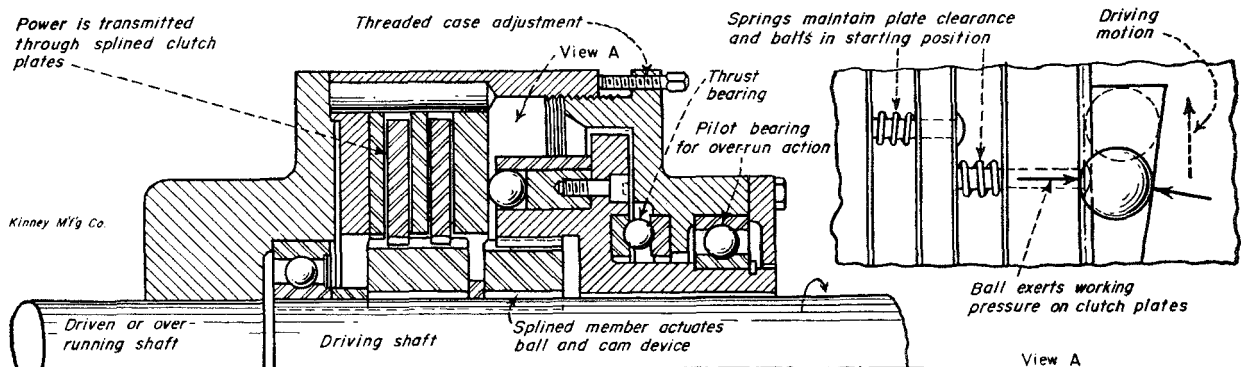
1 Elementary over-riding clutches: (A) Ratchet and Pawl mechanism is used to convert reciprocating or oscillating movement to intermittent rotary motion. This motion is positive but limited to a multiple of the tooth pitch. (B) Friction-

type is quieter but requires a spring device to keep eccentric pawl in constant engagement. (C) Balls or rollers replace the pawls in this device. Motion of the outer race wedges rollers against the inclined surfaces of the ratchet wheel.



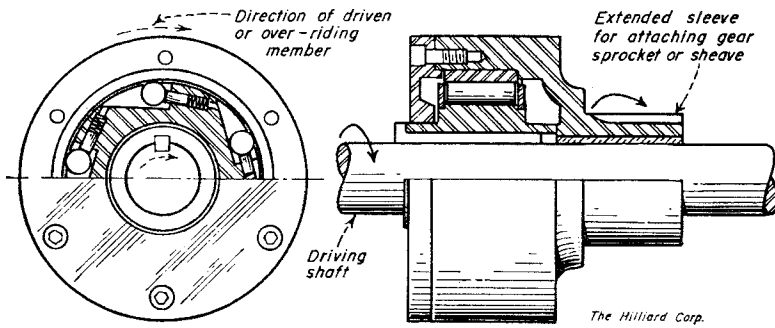
4 With cylindrical inner and outer races, sprags are used to transmit torque. Energizing springs serves as a cage to hold the sprags. (A) Compared to rollers, shape of sprag permits a greater number within a limited space; thus higher torque loads

are possible. Not requiring special cam surfaces, this type can be installed inside gear or wheel hubs. (B) Rolling action wedges sprags tightly between driving and driven members. Relatively large wedging angle insures positive engagement.

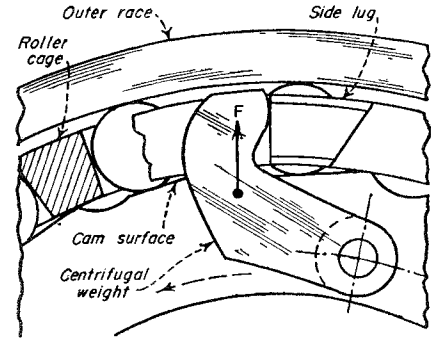


6 Multi-disk clutch is driven by means of several sintered-bronze friction surfaces. Pressure is exerted by a cam actuating device which forces a series of balls against a disk plate. Since a small part of the transmitted torque is carried by

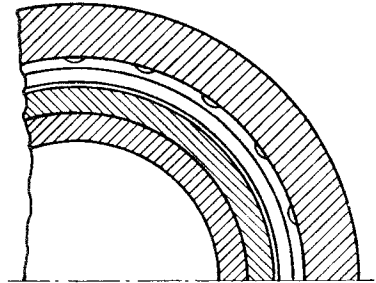
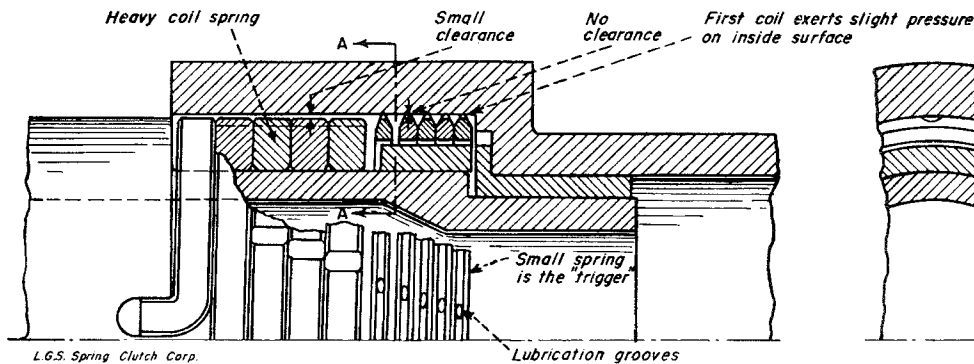
the actuating member, capacity is not limited by the localized deformation of the contacting balls. Slip of the friction surfaces determine the capacity and prevent rapid, shock loads. Slight pressure of disk springs insure uniform engagement.



2 Commercial over-riding clutch has springs which hold rollers in continuous contact between cam surfaces and outer race; thus there is no backlash or lost motion. This simple design is positive and quiet. For operation in the opposite direction, the roller mechanism can easily be reversed in the housing.

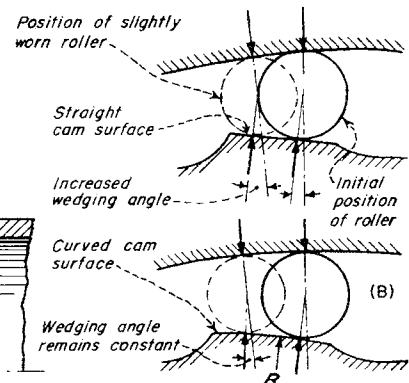
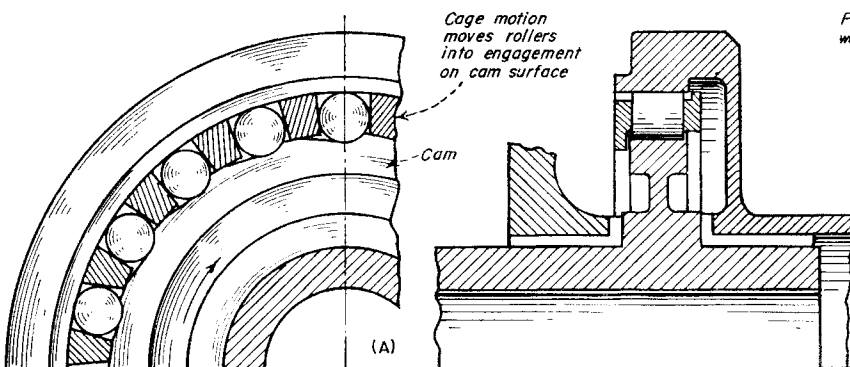


3 Centrifugal force can be used to hold rollers in contact with cam and outer race. Force is exerted on lugs of the cage which controls the position of the rollers.



5 Engaging device consists of a helical spring which is made up of two sections: a light trigger spring and a heavy coil spring. It is attached to and driven by the inner shaft. Relative motion of outer member rubbing on trigger causes this spring to

wind-up. This action expands the spring diameter which takes up the small clearance and exerts pressure against the inside surface until the entire spring is tightly engaged. Helix angle of spring can be changed to reverse the over-riding direction.



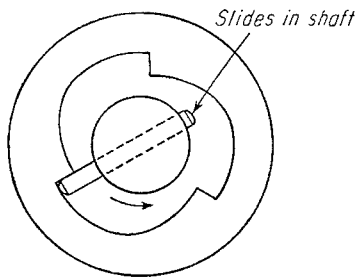
7 Free-wheeling clutch widely used in power transmission has a series of straight-sided cam surfaces. An engaging angle of about 3 deg is used; smaller angles tend to become locked and are difficult to disengage while larger ones are not as

effective. (A) Inertia of floating cage wedges rollers between cam and outer race. (B) Continual operation causes wear of surfaces; 0.001 in. wear alters angle to 8.5 deg. on straight-sided cams. Curved cam surfaces maintain constant angle.

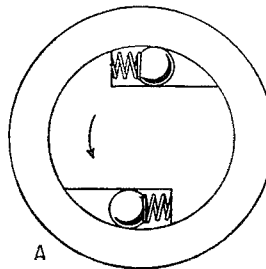
Low-Cost Designs for Overrunning Clutches

All are simple devices that can be constructed inexpensively in the laboratory workshop.

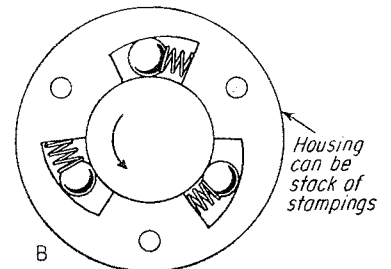
James F. Machen



1
Lawnmower type

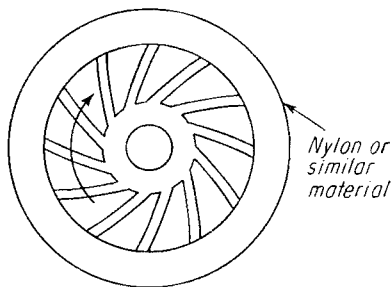


A

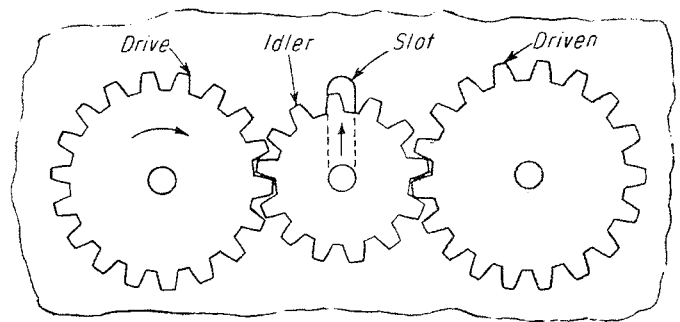


B

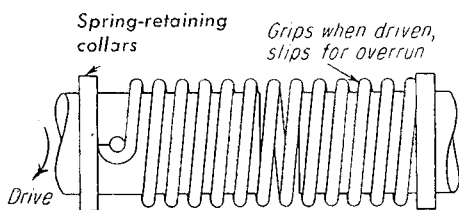
2
Wedging balls or rollers: internal (A); external (B)



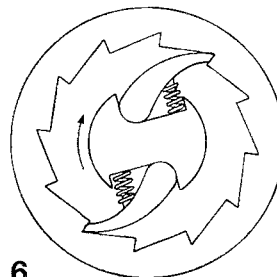
3
Molded sprags
(for light duty)



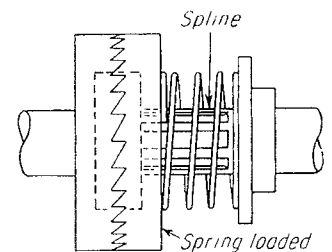
4
Disengaging idler rises in slot when drive direction is reversed



5
Slip-spring coupling

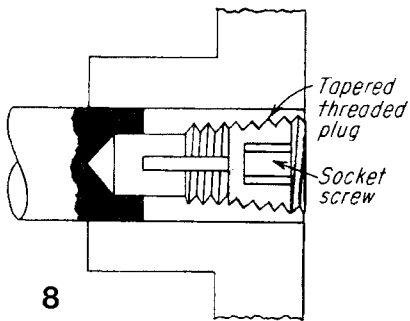


6
Internal ratchet and spring-loaded pawls

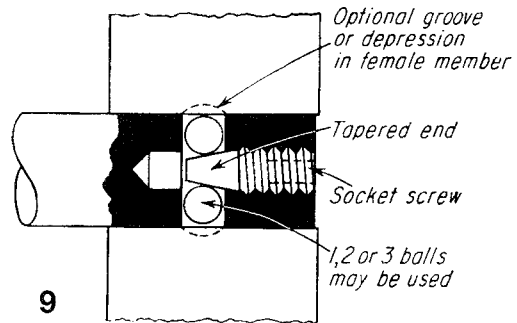


7
One-way dog clutch

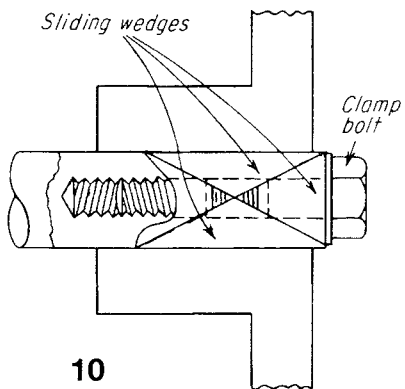
(FOR ANGULAR AND AXIAL ADJUSTMENT)



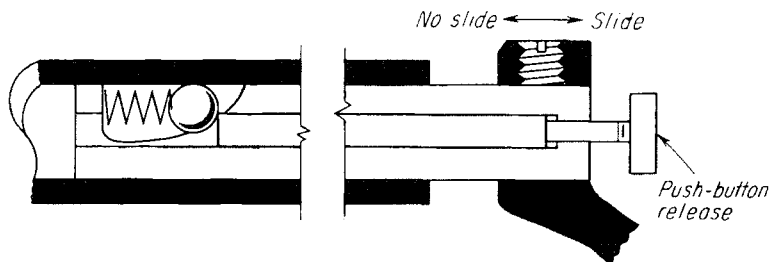
8
Plug jam
(with four axial slots)



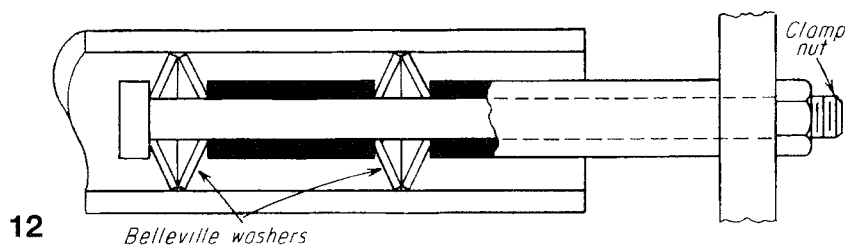
9
Ball jam



10
Clamped wedges
(for axial adjustment only)



11
One-way slide-lock

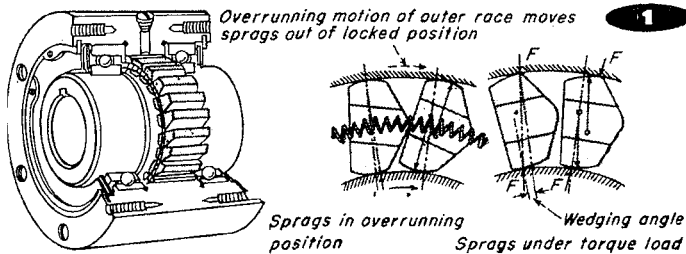


12
Belleville-washer clamp

10 Ways to Apply Overrunning Clutches

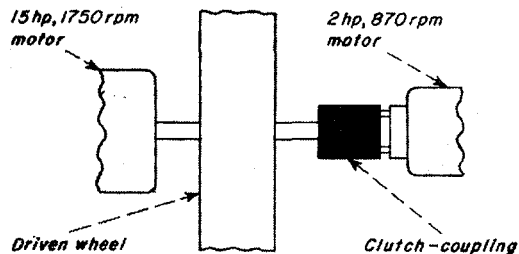
These clutches allow freewheeling, indexing, and backstopping applicable to many design problems. Here are some clutch setups.

W. Edgar Mulholland & John L. King JR.



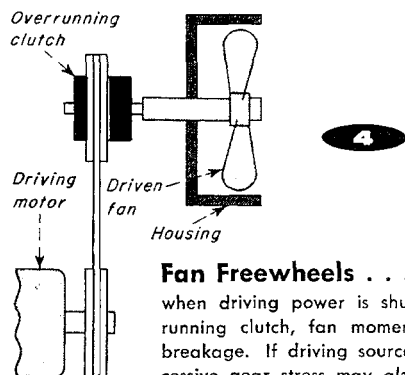
Precision Sprags . . .

act as wedges and are made of hardened alloy steel. In the Formsprag clutch, torque is transmitted from one race to another by wedging action of sprags between the races in one direction; in other direction the clutch freewheels.



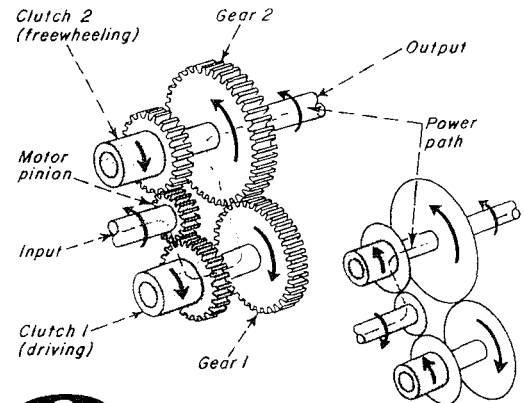
2-Speed Drive - II . . .

for grinding wheel can be simple, in-line design if over-running clutch couples two motors. Outer race of clutch is driven by gearmotor; inner race is keyed to grinding-wheel shaft. When gearmotor drives, clutch is engaged; when larger motor drives, inner race over-runs.



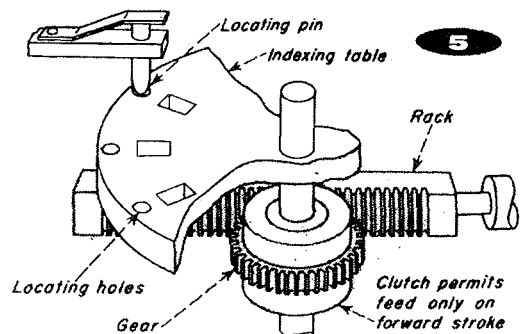
Fan Freewheels . . .

when driving power is shut off. Without over-running clutch, fan momentum can cause belt breakage. If driving source is a gearmotor, excessive gear stress may also occur by feedback of kinetic energy from fan.



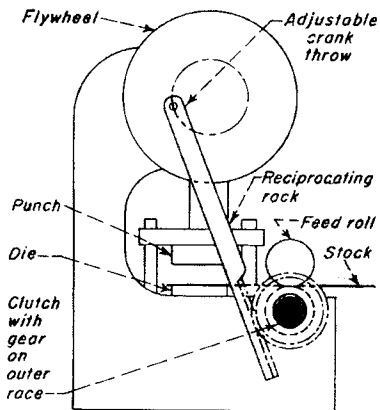
2-Speed Drive - I . . .

requires input rotation to be reversible. Counterclockwise input as shown in the diagram drives gear 1 through clutch 1; output is counterclockwise; clutch 2 over-runs. Clockwise input (schematic) drives gear 2 through clutch 2; output is still counterclockwise; clutch 1 over-runs.



Indexing Table . . .

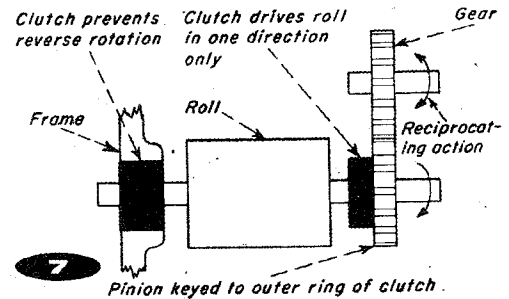
is keyed to clutch shaft. Table is rotated by forward stroke of rack, power being transmitted through clutch by its outer-ring gear only during this forward stroke. Indexing is slightly short of position required. Exact position is then located by spring-loaded pin, which draws table forward to final positioning. Pin now holds table until next power stroke of hydraulic cylinder



6

Punch Press Feed . .

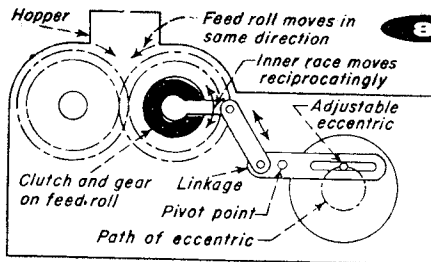
is so arranged that strip is stationary on down-stroke of punch (clutch freewheels); feed occurs during upstroke when clutch transmits torque. Feed mechanism can easily be adjusted to vary feed amount.



7

Indexing and Backstopping . . .

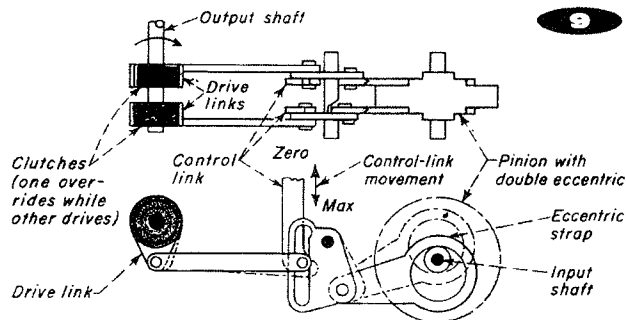
is done with two clutches so arranged that one drives while the other freewheels. Application here is for capsuling machine; gelatin is fed by the roll and stopped intermittently so blade can precisely shear material to form capsules.



8

Intermittent Motion . . .

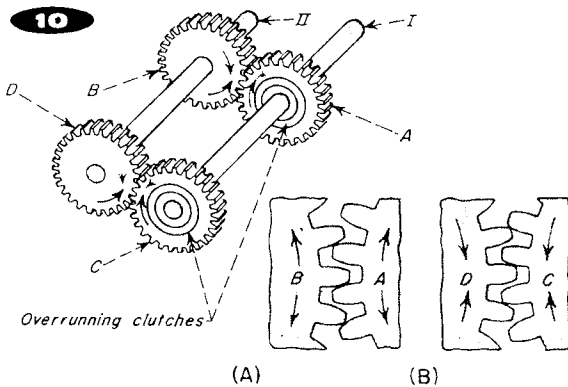
of candy machine is adjustable; function of clutch is to ratchet the feed rolls around. This keeps the material in the hopper agitated.



9

Double-impulse Drive . . .

employs double eccentrics and drive clutches. Each clutch is indexed 180° out of phase with the other. One revolution of eccentric produces two drive strokes. Stroke length, and thus the output rotation, can be adjusted from zero to max by the control link.



(A)

(B)

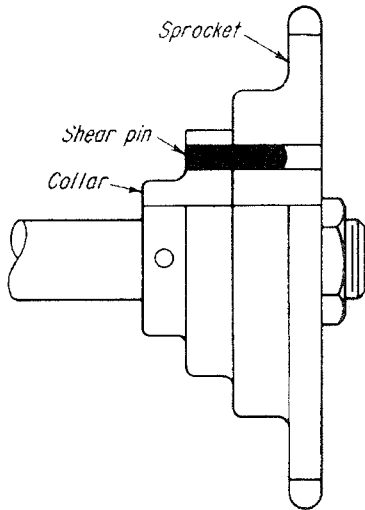
Anti-backlash Device . . .

uses over-running clutches to insure that no backlash is left in the unit. Gear A drives B and shaft II with the gear mesh and backlash as shown in (A). The over-running clutch in gear C permits gear D (driven by shaft II) to drive gear C and results in the mesh and backlash shown in (B). The over-running clutches never actually over-run. They provide flexible connections (something like split and sprung gears) between shaft I and gears A, C to allow absorption of all backlash.

6 Ways to Prevent Overloading

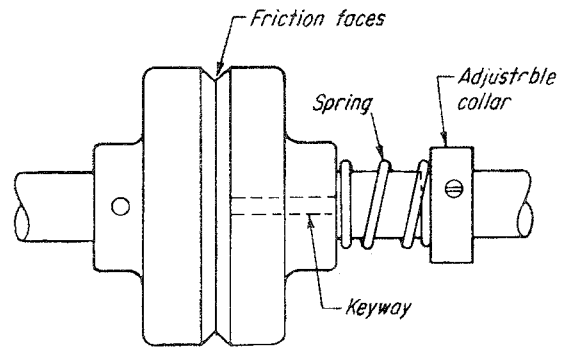
These "safety valves" give way if machinery jams, thus preventing serious damage.

Peter C. Noy

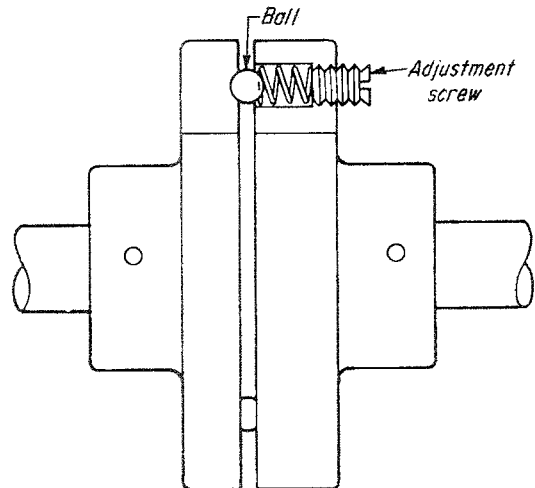


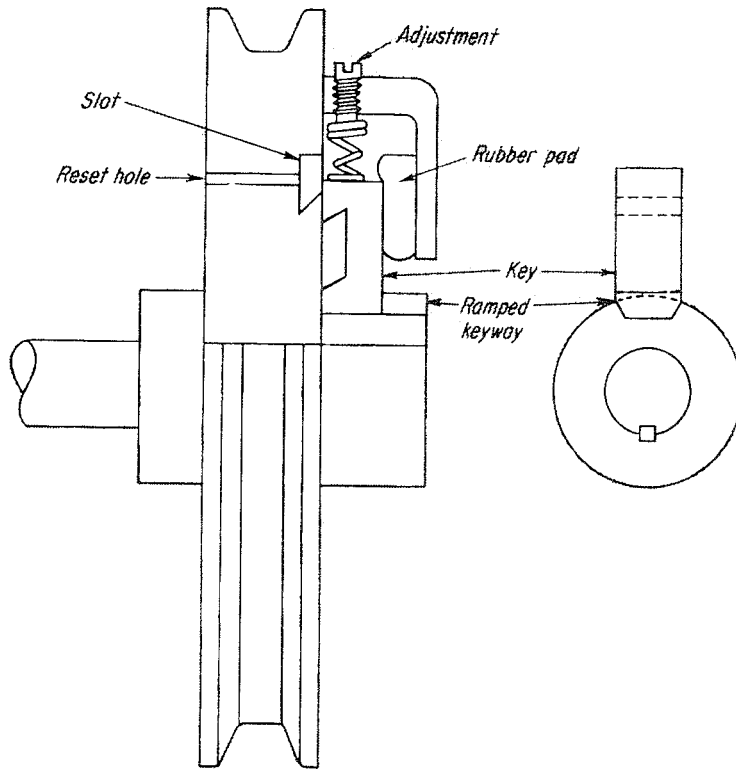
1
SHEAR PIN is simple to design and reliable in service. However, after an overload, replacing the pin takes a relatively long time; and new pins aren't always available.

3
MECHANICAL KEYS. Spring holds ball in dimple in opposite face until overload forces the ball out. Once slip begins, wear is rapid, so device is poor when overload is common.

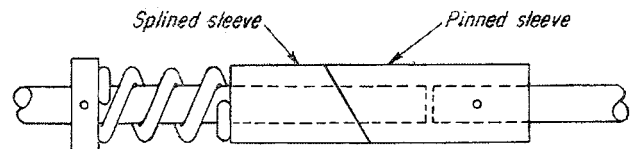
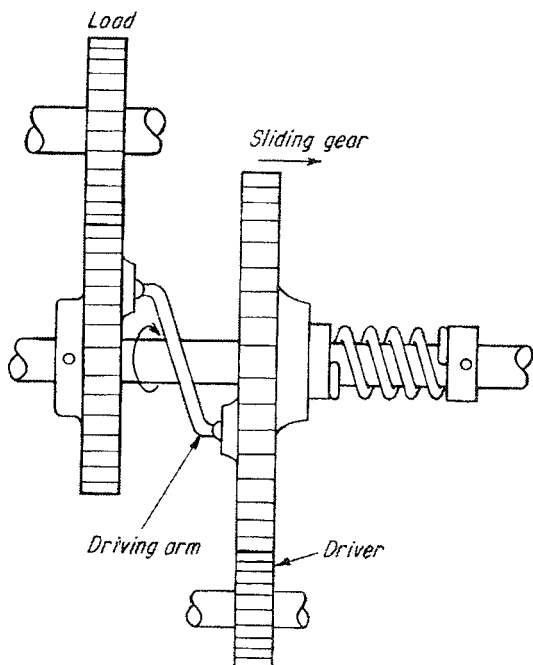


2
FRICITION CLUTCH. Adjustable spring tension that holds the two friction surfaces together sets overload limit. As soon as overload is removed the clutch reengages. One drawback is that a slipping clutch can destroy itself if unnoticed.



**4**

RETRACTING KEY. Ramped sides of keyway force key outward against adjustable spring. As key moves outward, a rubber pad—or another spring—forces the key into a slot in the sheave. This holds the key out of engagement and prevents wear. To reset, push key out of slot by using hole in sheave.

**5**

ANGLE-CUT CYLINDER. With just one tooth, this is a simplified version of the jaw clutch. Spring tension sets load limit.

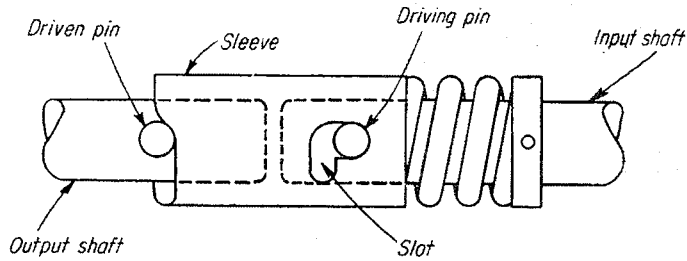
6

DISENGAGING GEARS. Axial forces of spring and driving arm balance. Overload overcomes spring force to slide gears out of engagement. Gears can strip once overloading is removed, unless a stop holds gears out of engagement.

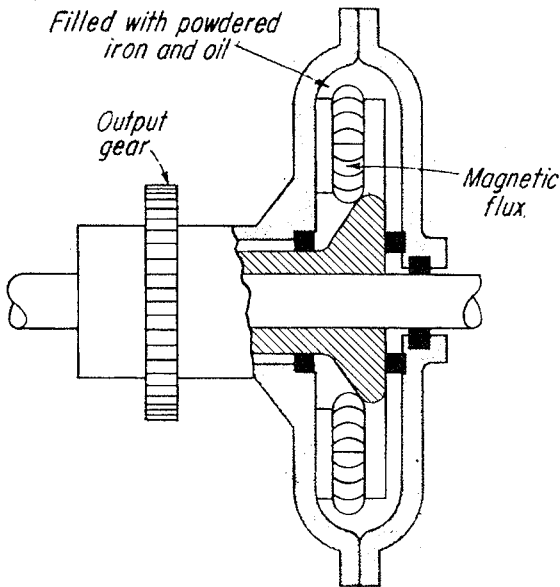
7 More Ways to Prevent Overloading

For the designer who must anticipate the unexpected, here are ways to guard machinery against carelessness or accident.

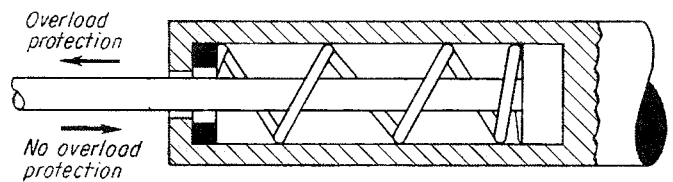
Peter C. Noy



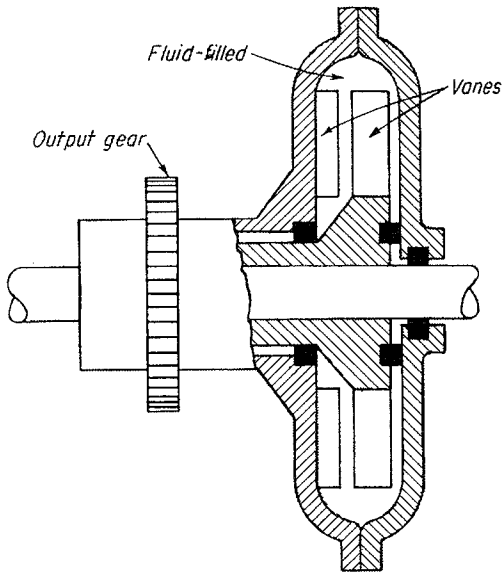
1
CAMMED SLEEVE connects input and output shafts. Driven pin pushes sleeve to right against spring. When overload occurs, driving pin drops into slot to keep shaft disengaged. Turning shaft backwards resets.



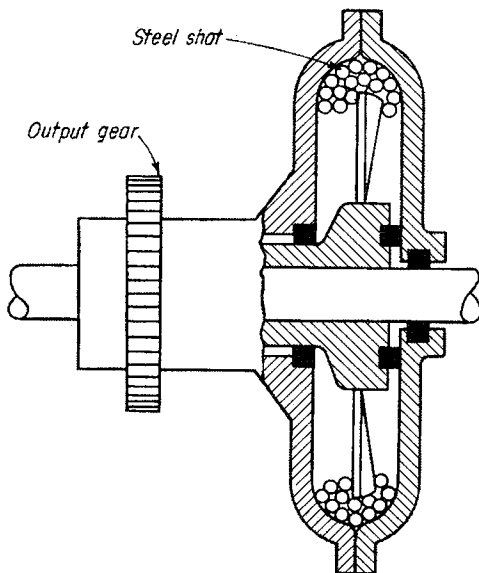
2
MAGNETIC FLUID COUPLING is filled with slurry made of iron or nickel powder in oil. Controlled magnetic flux that passes through fluid varies slurry viscosity, and thus maximum load over a wide range. Slip ring carries field current to vanes.



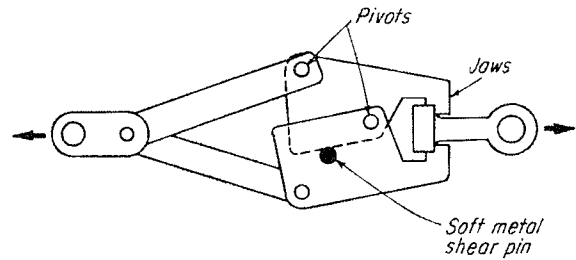
3
SPRING PLUNGER is for reciprocating motion with possible overload only when rod is moving left. Spring compresses under overload.



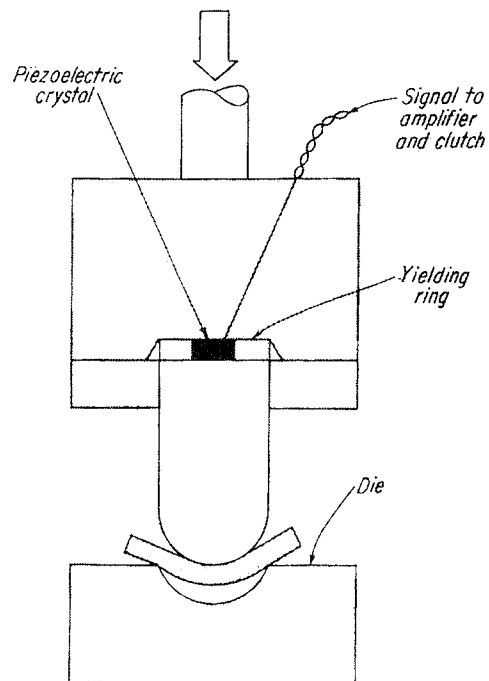
4
FLUID COUPLING. Maximum load can be closely controlled by varying viscosity and level of fluid. Other advantages are smooth transmission and low heat rise during slip.



6
STEEL-SHOT COUPLING transmits more torque as speed increases. Centrifugal force compresses steel shot against case, increasing resistance to slip. Adding more steel shot also increases resistance to slip.



5
TENSION RELEASE. When toggle-operated blade shears soft pin, jaws open to release eye. A spring that opposes the spreading jaws can replace the shear pin.



7
PIEZOELECTRIC CRYSTAL sends output signal that varies with pressure. Clutch at receiving end of signal disengages when pressure on the crystal reaches preset limit. Yielding ring controls compression of crystal.

Centrifugal Clutches

These simple devices provide low-cost clutching for machines operating under fast-changing load conditions.

M. F. Spotts

IF you want a practical way of connecting a motor—or any other type of prime mover—to a load when frequent stopping and starting are involved, centrifugal clutches are a good bet. Low in initial cost, centrifugal clutches can save you the expense of buying another form of electric motor, not to mention auxiliary starting equipment. Centrifugal clutches are built in a wide range of sizes and types—all the way from the little gasoline cars run on tracks at funfairs (here the motor disengages at low speed, and as the child presses on the gas pedal the car starts moving) to 500-hp diesel engines.

Other advantages: These clutches are good starters for high-inertia loads. They tolerate a considerable amount of manufacturing variations and are well suited to drives that undergo vibrations and heavy shock loads. Delayed engagements are possible by varying the clutch-spring force, and installation and service costs are low.

A typical centrifugal clutch has a set of shoes that are forced out against the output drum by centrifugal force. The shoes may be loosely held within the drum (Fig 1, next page), but in the more refined designs the shoes are connected to the input member by means of a floating link (Fig 2) or a fixed pivot (Fig 5).

Both attached-shoe designs (Fig 2 and 5) are analyzed here. Until now, however, the design procedure has been basically a graphical one. The design formulas derived here obviate the need for a graphical layout (or the graphical solution can serve as a check).

Floating-link design

The shoe of this type (Fig 2) is supported at the free end H of the floating link BH . End B of the link is attached to the hub of the driving member. The lining contacts the drum of the driven member and covers an angle ϕ , with support H at the midpoint which determines the v and w axes for the shoe. Angle ϕ is large so pressure p between the lining and drum is not constant but varies according to the equation

$$p = p_o \cos (\psi - \theta) \quad (1)$$

where p_o is the maximum value of the lining pressure located at angle θ to the v -axis. Angle ψ is the angle from the v -axis to the element under consideration.

The pressure on the lining p , when multiplied by the

area $br d\psi$ of an element of lining, gives the normal force between lining and drum. The component of force parallel to the v -axis is found by multiplying by $\cos \psi$. Eq 1 is substituted for p and the result integrated over the length of the lining, $-\phi/2$ to $\phi/2$, to obtain N_v , the component of normal force parallel to v -axis:

$$N_v = - \int_{-\phi/2}^{\phi/2} pbr \cos \psi d\psi$$

$$N_v = -\frac{1}{2}brp_o \cos \theta (\phi + \sin \phi) \quad (2)$$

In a similar manner, the total component of normal force N_w parallel to the w -axis is found by multiplying by $\sin \psi$. The integration gives

$$N_w = - \int_{-\phi/2}^{\phi/2} pbr \sin \psi d\psi$$

$$N_w = -\frac{1}{2}brp_o \sin \theta (\phi - \sin \phi) \quad (3)$$

The accompanying friction forces are

$$F_v = \int \mu pbr \sin \psi d\psi = -\mu N_w \quad (4)$$

$$F_w = - \int \mu pbr \cos \psi d\psi = \mu N_v \quad (5)$$

where μ is the coefficient of friction.

The torque exerted by the shoe is found by multiplying the normal force on the element by μ , and multiplying again by r to give:

Torque equation

$$T = \int_{-\phi/2}^{\phi/2} \mu pbr^2 d\psi = 2\mu br^2 p_o \cos \theta \sin \frac{\phi}{2} \quad (6)$$

(Eq 2 to 6 were obtained by G. A. G. Fazekas, "Graphical Shoe-Brake Analysis," Trans. ASME, vol 79, 1957, p 1322.)

The clutch shoe is in equilibrium from the following three forces:

1. An outward radial force consisting of the difference between the centrifugal force F_c on the shoe and the inward force F_s from the springs.

2. Force Q which is the resultant of the previously determined forces N_v , N_w , F_v and F_w .

3. Reaction R which must have a BH direction since this is a two-force member.

When a body with three forces is in equilibrium, the three forces must intersect at a single point. Because R and $F_c = F_s$ intersect at H , force Q must pass through this point also. Fig 3 shows Q passing through H as the resultant of the other two forces.

Since Q is also the resultant of the forces arising from p and μp , the moment made by such forces about H must be zero. This fact is utilized for finding θ , the inclination of the line of maximum pressure. The moment arms about H for p and μp are marked in the figure. Then

$$\int p b r h \sin \psi d\psi - \int \mu p b r (r - h \cos \psi) d\psi = 0$$

The value of the variable p from Eq 1 is substituted, and the resulting expressions are integrated between the limits of $-\phi/2$ and $\phi/2$. The result can be solved for $\tan \theta$ to give:

Line of maximum pressure

$$\tan \theta = \frac{4\mu r \sin \frac{\phi}{2} - \mu h (\phi + \sin \phi)}{h (\phi - \sin \phi)} \quad (7)$$

When θ is determined, the forces represented by Eq 2 to 5 inclusive can be found and added vectorially as shown in Fig 3. This gives:

Resultant of forces, Q

$$Q = \sqrt{(N_v + F_v)^2 + (N_w + F_w)^2} \quad (8)$$

Inclination β of force Q

$$\tan \beta = \frac{N_w + F_w}{N_v + F_v} \quad (9)$$

At low or idling speeds, the shoe is pulled inwardly

by the spring force F_s . The outward force F_c on the shoe is equal to:

Centrifugal force

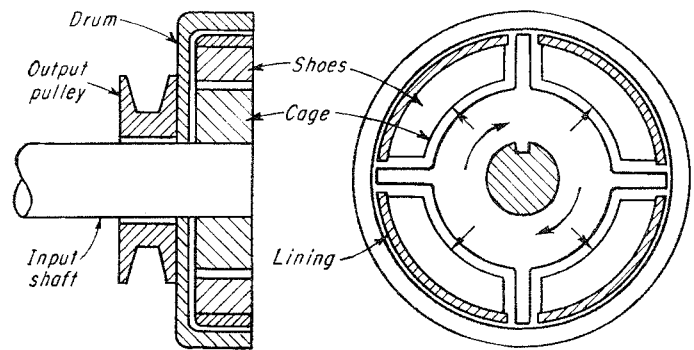
$$F_c = \frac{W}{g} r_o \omega^2 \quad (10)$$

where W is the weight of the shoe assumed to be concentrated at radius r_o ; ω is the angular velocity, rad/sec, and g is the gravitational constant, 386 in./sec².

As shown in Fig 2, link BH is inclined at angle α to the v -axis. The sine theorem gives equations for:

Forces R and $F_c = F_s$

$$R = \frac{Q \sin \beta}{\sin \alpha} \quad (11)$$



1. Free-shoe centrifugal clutch

SYMBOLS

b = width of lining

F_c = centrifugal force

F_s = spring force

F_v = component along v -axis of friction lining force

F_w = component along w -axis of friction lining force

g = gravitational constant, 386 in./sec²

h = distance from center of rotation to pivot of shoe

hp = transmittable horsepower

M_f = moment of friction forces about fixed pivot

M_n = moment of normal forces about fixed pivot

n = revolutions per minute

N_v = component along v -axis of forces normal to lining

N_w = component along w -axis of forces normal to lining

p = pressure between lining and drum

p_o = maximum lining pressure

Q = resultant of N_v , N_w , F_v and F_w

r = radius of drum

r_o = radius to center of gravity of shoe; outer radius of fixed pivot shoe

r_i = inner radius of fixed pivot shoe

r_p = distance from center of rotation to fixed pivot

R = reaction along floating link

T = torque about center of rotation

W = weight of shoe

α = inclination of floating link with v -axis

β = inclination of resultant Q with v -axis

γ = weight per in.³

θ = inclination of line of maximum pressure with v -axis

μ = coefficient of friction

ϕ = angular extent of lining

ω = angular velocity before slip occurs, radians/sec

$$F_c - F_s = \frac{Q \sin(\alpha - \beta)}{\sin \alpha} \quad (12)$$

Numerical example

A centrifugal clutch shoe has a radius of 5.25 in. and a width of lining of 2.50 in. Lining pressure is not to exceed 100 psi. Angular length of lining is $\phi = 108$ deg, and the link is at an angle of $\alpha = 48$ deg. The shoe is pivoted at a distance h of 4 in. from the center. Total inward spring force of both springs is 15 lb. Weight of shoe is 3 lb with its center of gravity at a radius of 4.6 in.

As there are linings on the market with a different coefficient of friction, for values of μ of 0.1, 0.2, 0.3, 0.4 and 0.5 find the corresponding values of forces Q , R , and $F_c - F_s$. Find the torque T , the corresponding

slipping rpm, and the horsepower per shoe. Plot the curve for hp vs μ to determine the best value for μ .

Solution

By Eq (7)

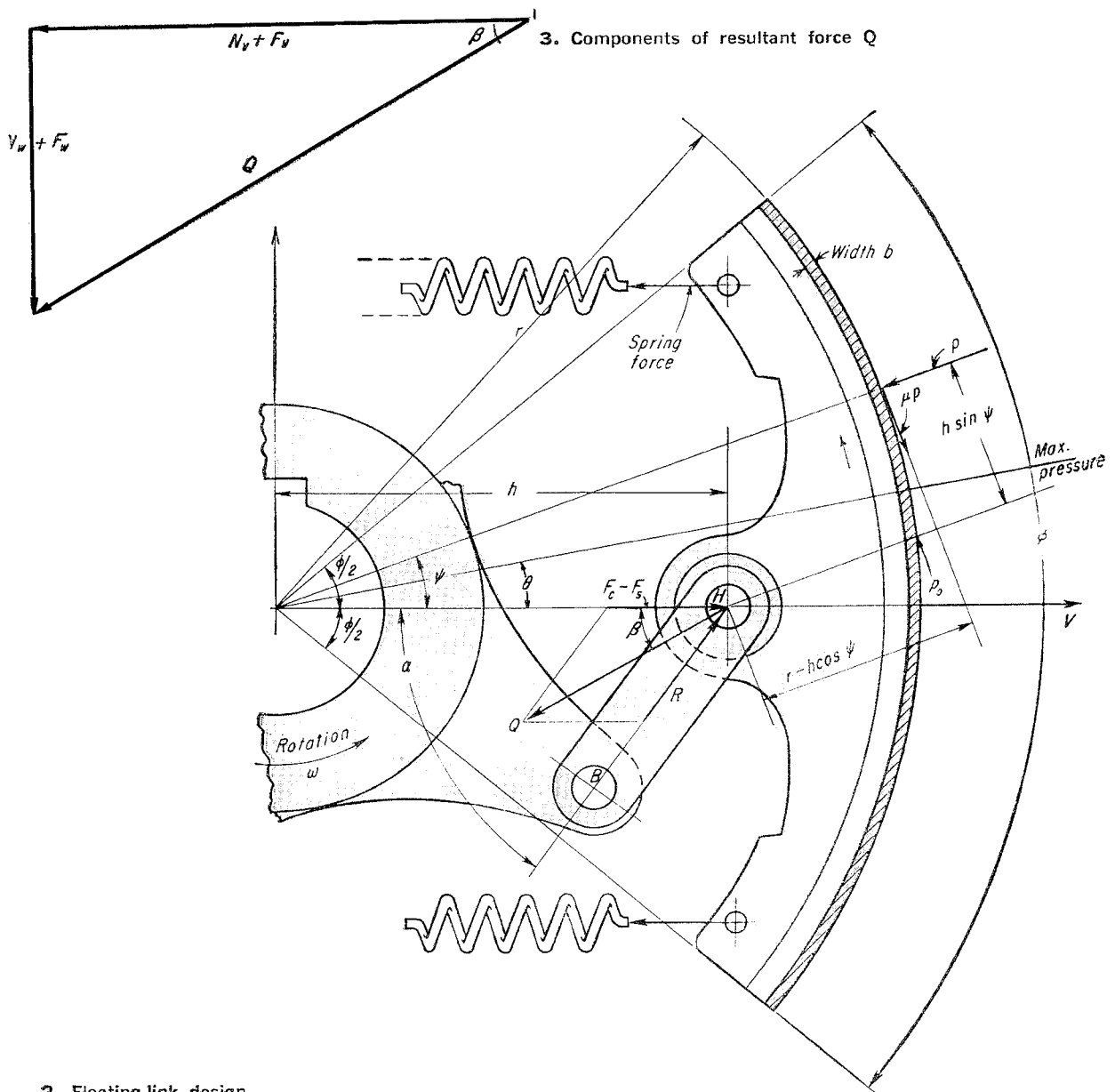
$$\tan \theta = \frac{\mu [(4)(5.25)(0.80902) - 4(1.88496 + 0.95106)]}{4(1.88496 - 0.95106)} = 1.511\mu$$

By Eq (2)

$$N_v = -\left(\frac{1}{3}\right)(2.5)(5.25)(100)(2.83602) \cos \theta = -1861 \cos \theta$$

By Eq (3)

$$N_w = -612.9 \sin \theta$$



By Eq (6)

$$T = 2\mu (2.5)(5.25)^2(100)(0.80902) \cos \theta \\ = 11,149 \mu \cos \theta$$

From Eq (10)

$$\omega = \sqrt{\frac{g F_c}{W r_o}} = \sqrt{\frac{386 F_c}{(3) 4.6}} = 5.29 \sqrt{F_c}$$

Then

$$n = \frac{\omega}{2\pi} 60 = 9.55 \omega = 50.5 \sqrt{F_c}$$

also

$$hp = \frac{Tn}{63,000}$$

The calculations are best carried out in tabular form (see table). Thus, for the case of $\mu = 0.1$

$$\tan \theta = 0.1511$$

$$\theta = 8^\circ 35.6'; \quad \cos \theta = 0.9888$$

$$N_v = -1861 (0.9888) = -1840.2 \text{ lb}$$

$$N_w = -612.9 (0.1494) = -91.6 \text{ lb}$$

By Eq 4, 5, and 8

$$F_v = -0.1 (-91.6) = 9.16 \text{ lb}$$

$$F_w = 0.1 (-1840.2) = -184.0 \text{ lb}$$

$$Q = \sqrt{(-1840.2 + 9.16)^2 + (-91.6 - 184.0)^2} \\ = 1852 \text{ lb}$$

By Eq 9

$$\tan \beta = \frac{-275.6}{-1831} = 0.15052$$

$$\beta = 8^\circ 33.6'$$

By Eq 11 and 12

$$R = \frac{1852 (\sin 8^\circ 33.6')}{\sin 48^\circ} = 371 \text{ lb}$$

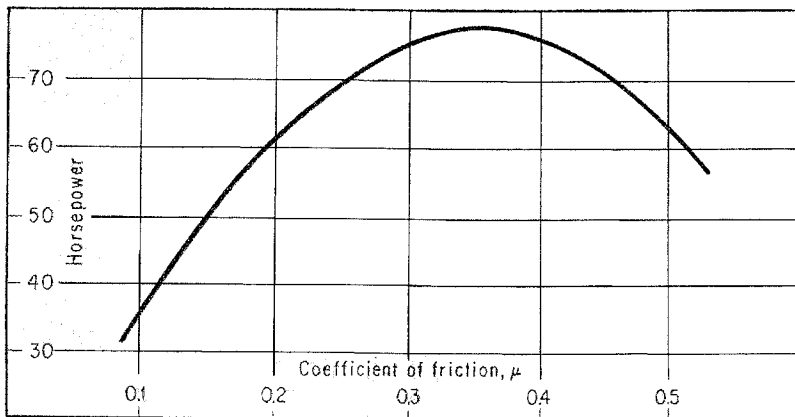
$$F_c - F_s = \frac{1852 \sin (48^\circ - 8^\circ 33.6')}{\sin 48^\circ} = 1583 \text{ lb}$$

$$F_c = 1583 + 15 = 1598 \text{ lb}$$

Thus

$$T = 11,149 (0.1) (0.9888) = 1102 \text{ lb-in.}$$

$$n = 50.5 \sqrt{1598} = 2019 \text{ rpm}$$



4. Variation of horsepower with coefficient of friction for the floating-link clutch analyzed in the numerical example. Note that best gripping power is obtained with shoe-linings having a coefficient of about 0.35%.

Calculations for various linings

μ	$\tan \theta$	θ	N_v	N_w	F_v	F_w	$N_v + F_v$	$N_w + F_w$	$(N_v + F_v)^2$	Q
0.1	0.15112	$8^\circ 35.6'$	-1840.2	-91.6	9.2	-184.0	-1831.0	-275.6	75,960	1852
0.2	0.30225	$16^\circ 49.0'$	-1774.1	-177.3	35.5	-354.8	-1738.6	-532.1	283,130	1818
0.3	0.45337	$24^\circ 23.3'$	-1695.1	-253.1	75.9	-508.5	-1619.2	-761.6	580,030	1789
0.4	0.60449	$31^\circ 9.2'$	-1592.7	-317.1	126.8	-637.1	-1465.9	-954.2	910,500	1749
0.5	0.75562	$37^\circ 4.5'$	-1484.9	-369.5	184.7	-742.4	-1300.2	-1111.9	1,236,320	1711
μ	$\tan \beta$	β	R	$\alpha - \beta$	$\sin (\alpha - \beta)$	$F_c - F_s$	T	F_c	n, rpm	hp
0.1	0.15052	$8^\circ 33.6'$	371	$39^\circ 26.4'$	0.63527	1583	1102	1598	2019	35.3
0.2	0.30605	$17^\circ 1.0'$	716	$30^\circ 59.0'$	0.51478	1259	2126	1274	1803	60.8
0.3	0.47036	$25^\circ 11.4'$	1024	$22^\circ 48.6'$	0.38768	933	3046	948	1556	75.2
0.4	0.65093	$33^\circ 3.7'$	1284	$14^\circ 56.3'$	0.25795	607	3817	622	1260	76.3
0.5	0.85518	$40^\circ 32.2'$	1496	$7^\circ 27.8'$	0.12990	299	4448	314	1011	63.2

Small Mechanical Clutches for Precise Service

Clutches used in calculating machines must have: (1) Quick response—lightweight moving parts; (2) Flexibility—permit multiple members to control operation; (3) Compactness—for equivalent capacity positive clutches are smaller than friction; (4) Dependability; and (5) Durability.

Marvin Taylor

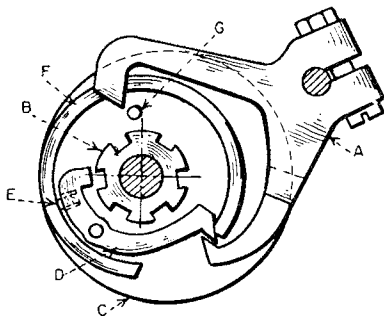


Fig. 1

PAWL AND RATCHET SINGLE CYCLE CLUTCH (Fig. 1). Known as Dennis Clutch, parts B, C and D, are primary components, B, being the driving ratchet, C, the driven cam plate and, D, the connecting pawl carried by the cam plate. Normally the pawl is held disengaged by the lower portion of clutch arm A. When activated, arm A rocks counter-clockwise until it is out of the path of rim F on cam plate C and permits pawl D under the effect of spring E to engage with ratchet B. Cam plate C then turns clockwise until, near the end of one cycle, pin G on the plate strikes the upper part of arm A camming it clockwise back to its normal position. The lower part of A then performs two functions: (1) cams pawl D out of engagement with the driving ratchet B and (2) blocks further motion of rim F and the cam plate.

PAWL AND RATCHET SINGLE CYCLE DUAL CONTROL CLUTCH—(Fig. 2). Principal parts are: driving ratchet, B, directly connected to the motor and rotating freely on rod A; driven crank, C, directly connected to the main shaft of the machine and also free on A; and spring loaded ratchet pawl, D, which is carried by crank, C, and is normally held disengaged by latch E. To activate the clutch, arm F is raised, permitting latch E to trip and pawl D to engage with ratchet B. The left arm of clutch latch C, which is in the path of the lug on pawl D, is normally permitted to move out of interference by the rotation of the camming edge of crank C. For certain operations block H is temporarily lowered, preventing motion of latch C, resulting in disengagement of the clutch after part of the cycle until subsequent raising of block H permits motion of latch C and resumption of the cycle.

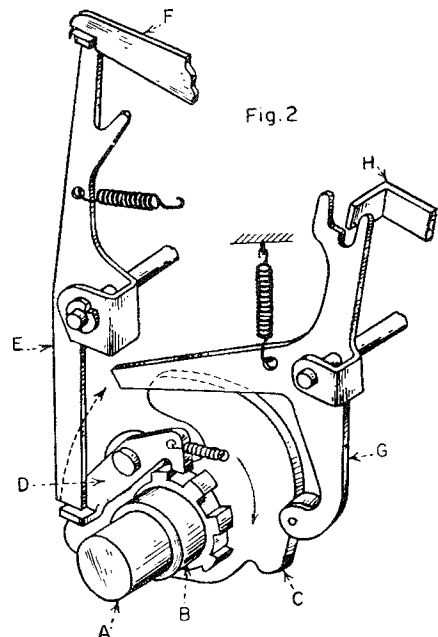


Fig. 2

PLANETARY TRANSMISSION CLUTCH (Fig. 3). A positive clutch with external control, two gear trains to provide bi-directional drive to a calculator for cycling the machine and shifting the carriage. Gear A is the driver, gear L the driven member is directly connected to planet carrier F. The planet consists of integral gears B and C; B meshing with sun gear A and free-wheeling ring gear G, and C meshing with free-wheeling gear D. Gears D and G carry projecting lugs, E and H respectively, which can contact formings on arms J and K of the control yoke. When the machine is at rest, the yoke is centrally positioned so that the arms J and K are out of the path of the projecting lugs permitting both D and G to free-wheel. To engage the drive, the yoke rocks clockwise as shown, until the forming on arm K engages lug H blocking further motion of ring gear G. A solid gear train is thereby established driving F and L in the same direction as the drive A and at the same time altering the speed of D as it continues counter-clockwise. A reversing signal rotates the yoke counter-clockwise until arm J encounters lug E blocking further motion of D. This actuates the other gear train of the same ratio.

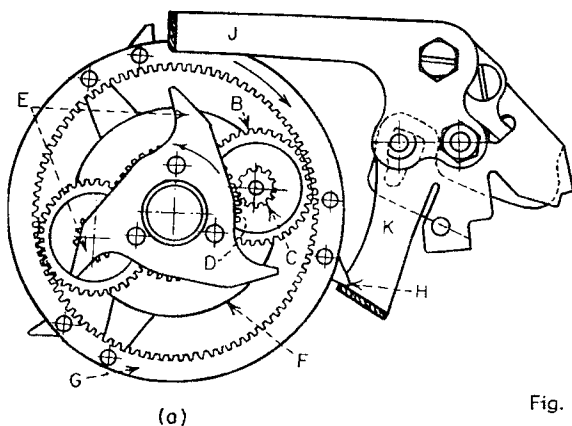
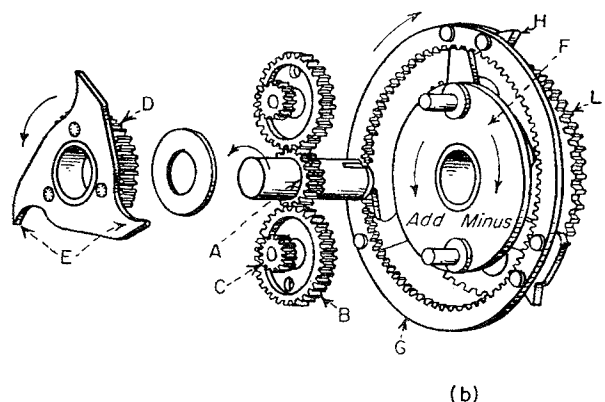


Fig. 3



(b)

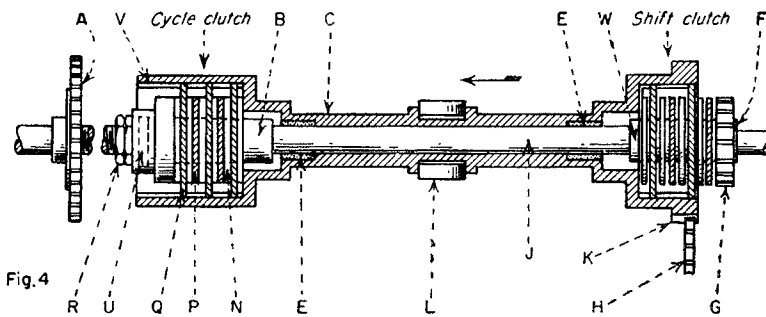


Fig. 4

MULTIPLE DISK FRICTION CLUTCH (Fig. 4). Two multiple disk friction clutches are combined in a single two-position unit which is shown shifted to the left. A stepped cylindrical housing C enclosing both clutches is carried by self-lubricated bearing E on shaft J and is driven by the transmission gear H meshing with the housing gear teeth K. At either end, the housing carries multiple metal disks Q that engage keyways V and can make frictional contact with formica disks N which, in turn, can contact a set of metal disks P which have slotted openings for coupling with flats on sleeves B and W. In the position shown, pressure is exerted through rollers L forcing the housing to the left making the left clutch compact against adjusting nuts R, thereby driving gear A via sleeve B which is connected to jack shaft J by pin U. When the carriage is to be shifted, rollers L force the housing to the right, first relieving the pressure between the adjoining disks on the left clutch then passing through a neutral position in which both clutches are disengaged and finally making the right clutch compact against thrust bearing F, thereby driving gear C through sleeve W which rotates freely on the jack shaft.

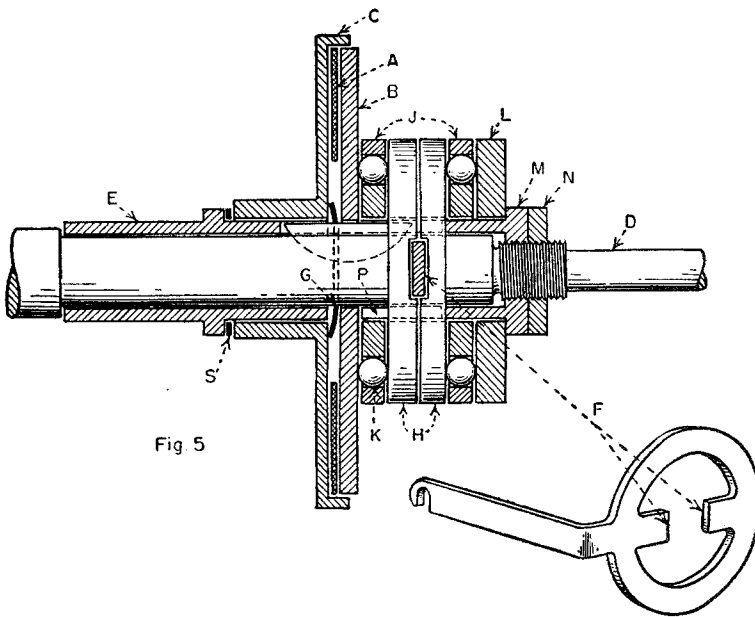


Fig 5

SINGLE PLATE FRICTION CLUTCH (Fig. 5). The basic clutch elements, formica disk A, steel plate B and drum C, are normally kept separated by spring washer G. To engage the drive, the left end of a control arm is raised, causing cars F, which sit in slots in plates H, to rock clockwise spreading the plates axially along sleeve P. Sleeves E and P and plate B are keyed to the drive shaft; all other members can rotate freely. The axial motion loads the assembly to the right through the thrust ball bearings K against plate L and adjusting nut M, and to the left through friction surfaces on A, B and C to thrust washer S, sleeve E and against a shoulder on shaft D, thus enabling plate A to drive the drum C.

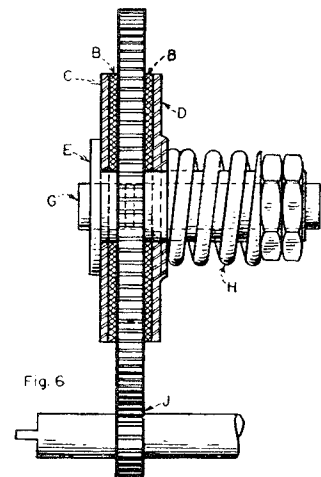
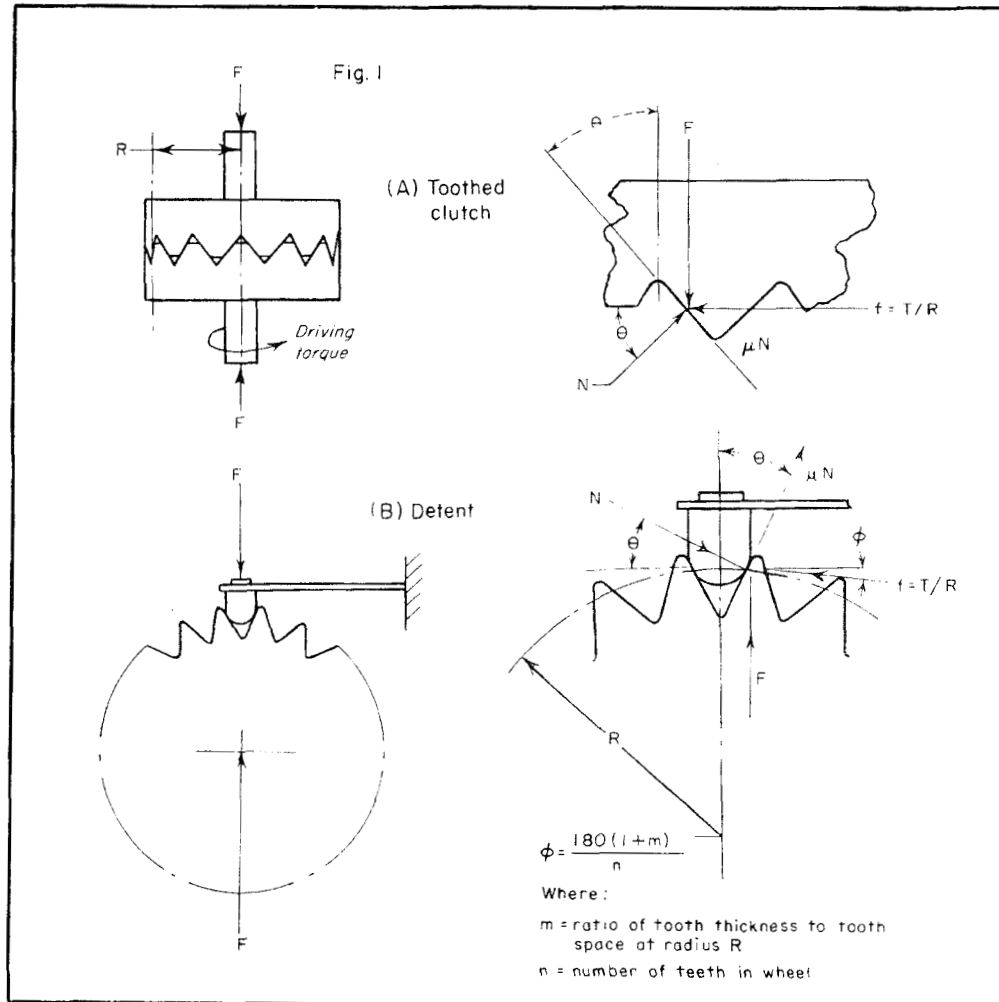


Fig 6

OVERLOAD RELIEF CLUTCH (Fig. 6). This is a simply constructed, double-plate, spring loaded, friction coupling. Shaft C drives collar E which drives slotted plates C and D and formica disks B. Spring H is forced by the adjusting nuts, which are screwed on to collar E, to maintain the unit under axial pressure against the shoulder at the left end of the collar. This enables the formica disks B to drive through friction against both faces of the gear which is free to turn on the collar, causing output pinion J to rotate. If the machine should jam and pinion J prevented from turning, the motor can continue running without overloading while slippage takes place between formica plates B and the gear.

Serrated Clutches and Detents

L. N. Canick



IN THE DESIGN OF straight toothed components such as serrated clutches, Fig. 1(A), and detent wheels, Fig. 1(B), the effective pitch radius is usually set by size considerations. The torque transmitting capacity of the clutch, or the torque resisting capacity of the detent wheel, is then obtained by assigning suitable values to the engaging force, tooth angle, and coefficient of friction.

The nomogram, Fig. 2, is designed to be a convenient means for considering the effect of variations in the values of tooth angle and coefficient of friction. For a given coefficient of friction, there is a tooth angle below which the clutch or detent is self-locking and will transmit torque limited only by its structural strength. Where

T = torque transmitted without clutch slip, or torque resisted by detent wheel, lb in.
 R = effective clutch, or detent wheel, radius, in.
 F = axial, or radial, force, lb
 f = tangential force acting at radius R , lb
 N = reaction force of driven tooth, or detent, acting normal to tooth face, lb

μ = coefficient of friction of tooth material
 θ = see Fig. 1(B)
 ϕ = angle of tooth face, deg
 $K = (1 + \mu \tan \theta) / (\tan \theta - \mu)$

a statement of the conditions of equilibrium for the forces acting on a clutch tooth will lead to the following equation

$$T = R F K \quad (1)$$

A similar statement of the conditions of equilibrium for the forces acting on a tooth of the detent wheel shown in Fig. 1(B) will lead to the following equation:

$$T = \frac{R F}{[(\cos \phi)/K] + \sin \phi} \quad (2)$$

From Eqs (1) and (2), when all other terms have constant values, it is obvious that the required axial force, or the radial force, diminishes as the value of K increases. Dependent upon the values of θ and μ , the value of K can vary from zero to infinity.

The circular nomogram shown in Fig. 2 relates the values of the parameters K , θ , and μ that satisfy the basic equation

$$K = (1 + \mu \tan \theta) / (\tan \theta - \mu)$$

EXAMPLE I. Find the maximum tooth angle for a self-locking clutch, or for which K is infinity, taking the coefficient of friction as 0.4 minimum.

SOLUTION I. Line I through these values for K and μ on the nomogram gives a maximum tooth angle slightly less than 22 deg for the self-locking condition.

EXAMPLE II. Find the minimum value of K to be expected for a clutch having a tooth angle of 30 deg and a coefficient of friction of 0.2 minimum.

SOLUTION II. Line II through these values for θ and μ on the nomogram gives a value for K of 3 approximately.

EXAMPLE III. Find the value of K for a flat-face (θ equals 90 deg) friction clutch, the face material of which has a coefficient of friction of 0.2. Compare its torque transmitting capacity with that of the toothed clutch of Example II.

SOLUTION III. Line III through these values for θ and μ on the nomogram gives a value for K of 0.2.

Torque transmitting capacity of flat-face clutch:

$$T = 0.2 R F$$

Torque transmitting capacity of toothed-clutch:

$$T = 3 R F$$

Thus for equal effective radii and engaging forces, the torque capacity of the toothed-clutch is $3/0.2$, or 15, times greater than that of the flat face clutch.

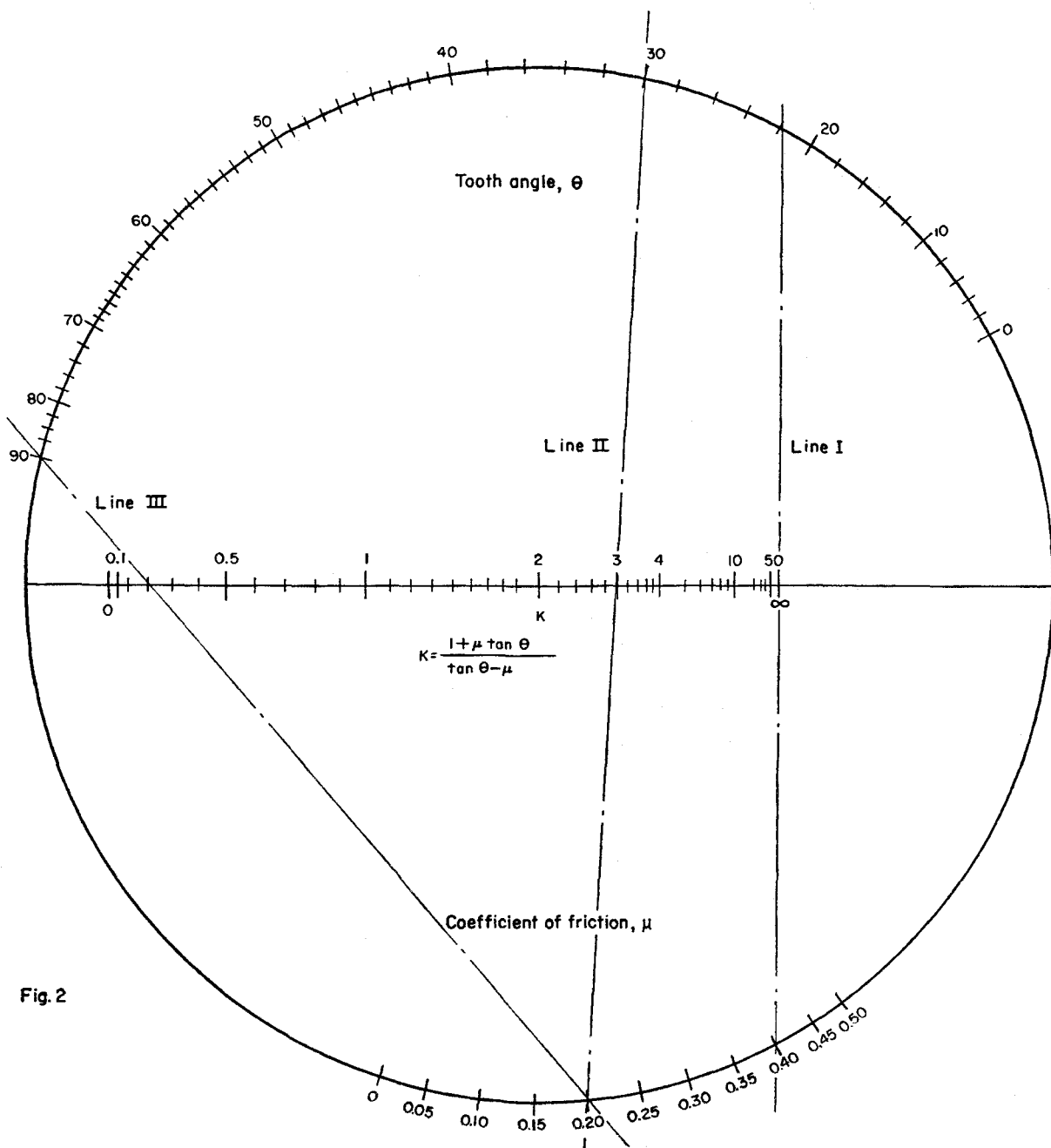
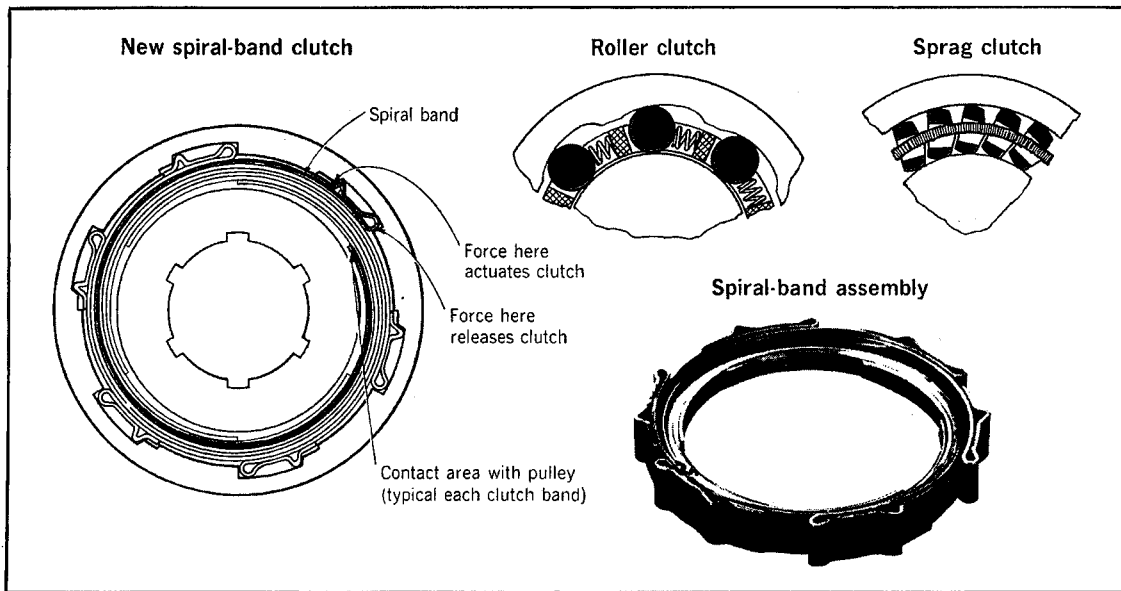


Fig. 2

Spring Bands Grip Tightly to Drive Overrunning Clutch



Spiral bands direct force inward as outer ring drives counterclockwise. Roller and sprag types direct force outward.

A new type of overrunning clutch that takes up only half the usual space employs a series of spiral-wound bands instead of the conventional rollers or sprags to transmit high torques. The new design (drawings, above) also simplifies the assembly, cutting costs as much as 40% by eliminating more than half the parts in conventional clutches.

The key to the savings in cost and bulk is the new design's freedom from the need for a hardened outer race. Roller and sprag types must have hardened races because they transmit power by a wedging action between the inner and outer races.

Role of spring bands. Overrunning clutches, including the spiral-band type, slip and overrun when reversed—in drawing above, when outer member is rotated clockwise and inner ring is the driven member.

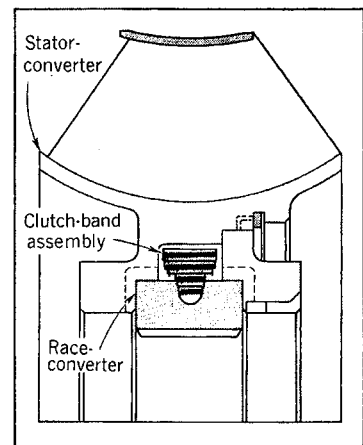
The new clutch, developed by National Standard Co., Niles, Mich., contains a set of high-carbon spring-steel bands (six in the design illustrated) that grip the inner member when the clutch is driving. The outer member merely serves to retain the spring anchors and to play

a part in actuating the clutch. Since it isn't subject to wedging action, it can be made of almost any material, and this accounts for much of the cost saving. For example, in the automotive torque converter in the drawing at right, the bands fit into the aluminum die-cast reactor.

Reduced wear. The bands are spring-loaded over the inner member of the clutch, but they are held and rotated by the outer member. The centrifugal force on the bands thus releases much of the force on the inner member and considerably decreases the overrunning torque. Wear is, therefore, greatly reduced.

The inner portion of the bands fits into a V-groove in the inner member. When the outer member is reversed, the bands wrap, creating a wedging action in this V-groove. This action is similar to that of a spring clutch with a helical-coil spring, but the spiral-band type has very little unwind before it overruns, compared with the coil type. Thus it responds faster.

Edges of the clutch bands carry the entire load, and there is also a compound action of one band upon



Spiral clutch bands can be bought separately to fit in user's assembly.

another. As the torque builds up, each band pushes down on the band beneath it, so each tip is forced more firmly into the V-groove.

National Standard plans to sell the bands as separate components, without the inner and outer clutch members (which the user customarily builds as part of his product). The bands are rated for torque capacities from 85 to 400 ft.-lb. Applications include auto transmissions and starters and industrial machinery. □

Accurate Solution for Disk-Clutch Torque Capacity

Nils M. Sverdrup

IN COMPUTING TORQUE CAPACITY, the mean radius R of the clutch disks is often used. The torque equation then assumes the following form:

$$T = P\mu Rn \quad (1)$$

Where

T = torque, in.-lb
 P = pressure, lb.
 μ = coefficient of friction
 R = mean radius of disks, in.
 n = no. of friction surfaces

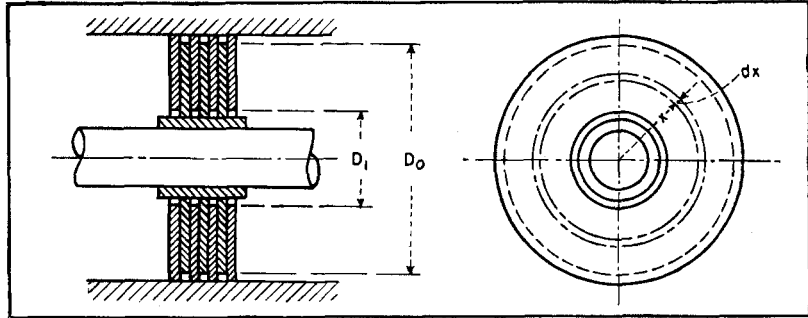
This formula, however, is not mathematically correct and should be used cautiously. The formula's accuracy varies with the ratio D_1/D_0 . When D_1/D_0 approaches unity, the error is negligible; but as the value of this ratio decreases, the induced error will increase to a maximum of 33 percent.

By introducing a correction factor, ϕ , Eq (1) can be written

$$T = P\mu Rn\phi \quad (2)$$

The value of the correction factor can be derived by the calculus derivation of Eq (2).

Sketch above represents a disk clutch with n friction surfaces, pressure between plates being p psi. Inside and outside diameters of effective friction areas are D_1 and D_0 in., respectively. Since the magnitude of pressure on an element of area, dA , at distance x from center is $p dA$, the



friction force is $p dA\mu$ and the moment of this force around the center is $p dA\mu x$.

Integrating within limits $D_1/2$ and $D_0/2$ and multiplying by n friction surfaces, the expression for total torque in in.-lb is obtained.

Hence

$$T = \int_{D_1/2}^{D_0/2} p dA \mu x n \quad (3)$$

but

$$dA = 2\pi x dx \quad (4)$$

Substituting in Eq (3)

$$\begin{aligned} T &= \int_{D_1/2}^{D_0/2} p (2\pi x dx) \mu x n \\ &= (2/3)\pi p \mu n \left[\left(\frac{D_0}{2}\right)^3 - \left(\frac{D_1}{2}\right)^3 \right] \\ \text{or } T &= 0.262 p \mu n (D_0^3 - D_1^3) \end{aligned} \quad (5)$$

If the total pressure acting on clutch disks be P lb, the expression for pressure per unit area is

$$p = \frac{P}{(\pi/4)(D_0^2 - D_1^2)}$$

Substituting this value for p in Eq (5)

$$T = 0.333 P \mu n \frac{D_0^2 + D_0 D_1 + D_1^2}{D_0 + D_1} \quad (6)$$

Now let

$$\frac{D_1}{D_0} = m \text{ so that, } D_1 = m D_0 \quad (7)$$

Substituting in Eq (6)

$$T = 0.333 P \mu n D_0 \frac{1 + m^2 + m}{1 + m} \quad (8)$$

Similarly, by substituting value of D_1 from Eq (7) in Eq (2), and having

$$R = \frac{D_0 + D_1}{4},$$

$$T = P \mu \frac{D_0 + m D_0}{4} n \phi$$

or

$$T = 0.25 P \mu n \phi D_0 (1 + m) \quad (9)$$

Equating expressions (8) and (9)

$$\begin{aligned} 0.25 P \mu n \phi D_0 (1 + m) &= \\ 0.333 P \mu n D_0 \frac{1 + m^2 + m}{1 + m} \end{aligned}$$

and solving for ϕ , the result is

$$\phi = 1.333 \times \frac{1 + m^2 + m}{(1 + m)^2} \quad (10)$$

With various diameter ratios, the values for ϕ were computed and represented in graph herewith. By using this graph and Eq (2), accurate values of torque can be easily determined.

